NUMERICAL SIMULATION OF HELICAL COIL TUBE IN TUBE HEAT EXCHANGER WITH BAFFLES

A REPORT SUBMITTED IN PARTIAL FULFILMENT

OF THE REQUIREMENTS FOR THE DEGREE OF

Master of Technology

In

Thermal Engineering

By

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Rourkela-769008, Odisha

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National Institute of Technology
Rourkela
CERTIFICATE

This is to certify that the thesis entitled, "**NUMERICAL SIMULATION OF TUBE-IN-TUBE HELICAL COIL HEAT EXCHANGER WITH BAFFLES**" submitted by Mr. **VISHNU M** bearing **Roll NO: 213ME3437** in partial fulfilment of the requirements for the award of Master of Technology Degree in Mechanical Engineering with specialization in Thermal Engineering at the National Institute of Technology, Rourkela (Deemed University) is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any degree or diploma.

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ABSTRACT

The project aims at studying the effect of providing baffles in helical coil tube in tube heat exchanger. A three dimensional helical coil tube in tube heat exchanger with number of turns equal to two and which consist of three ring shaped baffles which are placed in annular space between two helical tubes is considered for study. The main aim of providing baffle is to increase the turbulence and thereby increase the convection and the baffle also provides support and helps to maintain two coils concentric. The geometry was modelled using Ansys design modeler. The hot fluid flows through the inner tube and cold fluid flows through annular space. The fluid considered is water. The analysis is done using Ansys fluent. The heat transfer in helical coil heat exchanger is analysed by varying the flow rate of hot fluid and flow rate of cold fluid is kept constant.

The laminar and turbulent flow cases are studied. K–ε model is used to model the turbulence in the flow and the flow is analysed for counter flow heat exchanger setup. The variation of Nusselt number values with the change in Reynolds number of hot fluid is plotted. By varying the thickness of baffle in helical coil tube in tube heat exchanger the variation of Nusselt number with the variation of Reynolds number of hot fluid is studied. The coil diameter of the helical coil is also varied to study the effect for both laminar and turbulent flow. The D/d ratio is varied from 10 to 25 in steps of 5

Keywords: Helical coil heat exchanger, Baffles, Nusselt number

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ABBREVIATIONS

C_p =specific heat, J/kg-K

d_1=diameter of inner tube, mm

d_2=diameter of outer tube, mm

D = coil diameter, mm

D_h = hydraulic diameter, mm

t = Thickness of baffle mm

De =Dean Number

f = Fannings friction factor

G =mass velocity, m/s

h =heat transfer coefficient, W/m^2k

H =pitch of coil, mm

k =thermal conductivity, W/m-K

L =length of pipe, m

m =mass flow rate, kg/s

n= number of turns

V = Velocity, m/s

Nu_x =Local Nusselt Number

Nu_{avg} =Average Nusselt number

P= Pressure, N/m^2

q= heat flux, W/m2

Re =Reynolds Number

Re_{cr}= Critical Reynolds number

T = Temperature, K

Th_i=temperature of hot fluid at inlet, K

Th_o=temperature of hot fluid at outlet, K

Tc_i= temperature of cold fluid at inlet, K

Tc_o= temperature of cold fluid at outlet, K

T_w= wall temperature, K

T_f= fluid mean temperature, K

U_x, U_y and U_z = velocity in x, y, z directions

x, y, z coordinates

X, Y, Z body force in x, y, z directions

Greek symbol

ρ =density, Kg/m3

μ=dynamic viscosity, Kg/ms

Φ=Rayleigh dissipation factor

CHAPTER 1

INTRODUCTION

1.1. HEAT EXCHANGERS

Heat exchanger is that devices which are used for the transferring heat between different temperature fluids which may be directly in contact or may be flowing separately in two tubes or in two channels. Numerous applications of heat exchangers can be observed in our day today life, to say a few are condensers and evaporators used in refrigerators and air conditioners and in case of thermal power plant heat exchangers are used in, condenser, boilers, air coolers and chilling towers. In case of automobiles heat exchangers are in the form of radiators or in the form of oil coolers in engine. Large scale process industries and chemical industries use heat exchangers for the transferring heat between different temperature fluids which are single phase or two phase.

1.2. TYPE OF HEAT EXCHANGERS:

- Based on Heat transfer process

1. Direct Contact heat exchanger

In direct contact heat exchangers two immiscible fluids are directly mixed and heat transfer occurs between two fluids. The speciality of this type of heat exchangers are the absence of wall separating the hot fluid stream and cold fluid stream. The application of this kind of heat exchangers can be found in many places like in air conditioners, water cooling, humidifiers, industrial hot water heating and condensing plants.

2. Transfer Type of Heat Exchanger

In Transfer type of heat exchanger two fluid simultaneously flows through two tubes separated by walls. This are the most commonly used type heat exchanger due to simplicity in its construction

3. Regenerators type Heat Exchanger

A regenerative heat exchanger is that type of heat exchanger in which hot fluid heat is intermittently stored in a thermal storage medium and then it will be transferred to the cold fluid. To achieve this first the hot fluid is allowed to come in contact with the thermal storage medium which is usually the wall of heat exchanger and then the fluid is replaced with the cold fluid which will absorb the heat from the storage medium.

- Based on Constructional Features

1. Tubular Heat Exchanger

This type of heat exchangers consist of two concentric tubes in which one of the fluid flows through the inner tube and the second fluid flows through the annular space. Both the fluids are separated by the wall and heat transfer occurs through the walls

2. Shell and Tube Heat Exchanger

This type of heat exchanger consist of tube bundles which is set of tubes and a shell. The fluid which is to be heated or cooled is contained in one set of these tubes. The second fluid flows over the tubes that is to be heated or cooled in this way fluid can be either heated or absorb the heat required.

3. Finned tube Heat Exchanger

The principle which is in cooperated in this type of heat exchangers are that with the introduction of fin in the heat exchanger the heat transfer capacity of the heat exchanger can be improved. This is mainly used in gas to liquid type of heat exchanger and while using this fin is used in gas side.

4. Compact Heat Exchanger

A compact heat exchanger can be defined as heat exchanger which has area density (The ratio of the heat transfer surface area of a heat exchanger to its volume) for gas value is greater than 700 m^2/m^3 and for liquid or two-phase stream operation it is greater than 300 m^2/m^3. Compact heat exchanger are generally cross flow type where two fluid flow perpendicular to each other.

- Based on flow arrangement

1. Parallel Flow

In parallel flow heat exchangers the hot and cold fluid flows parallel to each other that means in the same direction.

2. Counter Flow

In counter flow heat exchangers both the fluids flows in opposite direction.

3. Cross Flow

In cross flow heat exchangers the two fluid flow perpendicular with respect to one another

1.3. HELICAL COIL HEAT EXCHANGERS

Helical coil heat exchanger are recent development which has many advantages when compared with straight tube heat exchangers.

Advantages:

a. Heat transfer rate of helical coil is large when compared with that of straight tube heat exchanger.
b. It has a compact structure and requires less floor area compared to other heat exchangers.
c. Self-cleaning.

d. Surface area for heat transfer is large

The application of heat exchanger covers following areas

1. Air conditioning
2. Power generation
3. Petroleum industry
4. Chilling towers in Thermal power plant
5. Refrigeration & cryogenics
6. For Heat recovery

1.4 OVERVIEW OF THE THESIS

This thesis is organized into six chapters

In chapter 1 I have tried to give brief introduction about the types of heat exchangers and various types of heat exchangers. Explanation about Helical coil, its advantage over straight tube heat exchanger and various applications of helical coil heat exchanger was also given

In chapter 2 I have tried to explain the various literature regarding helical coil which are surveyed and tried to explain the gaps in literature.

In chapter 3 I have tried to explain how i tried to formulate the problem and described the various cases that are studied.

In chapter 4 I have tried to explain how I have created the geometry and how meshing is done. The assumptions that are used in the problem and governing equations that are used to solve the problem.

In chapter 5 I have shown various results obtained and have interpreted the reasons for the various results obtained.

In chapter 6 I have described the various conclusions that are drawn and described the scope for future work

CHAPTER 2

LITERATURE REVIEW& GAPS IN LITERATURE

2.1. INTRODUCTION

In recent decade there was large growth in computing power and memory capacity which has led to increase interest for scientist, engineers and researchers to simulate their problem using computational and numerical methods. Many computational tools (software packages) and methods have been developed since the last decades to analyse many engineering problems related to fluid dynamics, combustion, and various modes of heat transfer. Heat exchangers are used for a large variety of applications which attract the researchers and scientists to do research related to heat exchangers.

The helical coil heat exchangers are widely used in many industries owing to its compact structural design, higher heat transfer capability and larger surface area for heat transfer. In the recent few years so many works has been done to improve the rate of heat transfer of heat exchangers. Wide range of literature has been found which shows that improvement in heat transfer rate can be achieved by using helical coil heat Exchanger. The secondary flow pattern of the fluid improves the heat transfer rate, the outer fluid moves with a higher velocity compared to the inner fluid due to the effect of curvature ratio. A large number of experimental work has been done to study flow pattern and also heat transfer characteristic of helical coil heat exchangers.

The works done on helical coil heat exchangers can be classified as follows

2.2. EXPERIMENTAL AND NUMERICAL WORKS

Naphon et al. (2005) has studied heat transfer characteristics of spiral coil heat exchanger which is subjected to wet-surface conditions, they have done both experimental and numerical studies to find out heat transfer rate as well as to predict spiral coil heat exchangers performance. For the analysis they used cooling and dehumidifying condition. The result that they got suggests that rate of mass flow and temperature of inlet air affects temperature of water and air at the outlet. The outlet temperature of air and water decrease with increase in water mass flow rate. With increase in mass flow rate of air and water rates the humidity effectiveness and enthalpy decrease.

Kumar et al. (2006) investigated heat transfer characteristics and hydrodynamics of tube in tube helical coil heat exchanger the experimental work was done on counter flow setup of heat exchanger and overall heat transfer coefficients was evaluated. The Nusselt number and friction coefficient for outer tube as well as inner tube was calculated and then it is compared with numerical values got from CFD software package FLUENT. The observation made by them is that overall heat transfer coefficient increase with inner coil dean number for constant flow rate in annulus region.

Jayakumar et al. (2008) has done both numerical and experimental study on helical coil heat exchanger and he has considered fluid to fluid heat transfer. The different boundary conditions under consideration where constant heat flux, constant wall temperature and constant heat transfer coefficient .The observation made was constant values of transport properties and thermal properties of heat transfer medium results in inaccurate heat transfer coefficient and also in many practical applications such as heat transfer in fluid to fluid heat exchangers arbitrary boundary conditions like constant heat flux and constant wall temperature are practically not applicable. Based on numerical analysis and experimental work conducted and development of correlations was done to calculate the inner heat transfer coefficient of helical coil within certain error limit.

Kharat et al. (2009) has done experiment on concentric helical coil heat exchanger to study the heat transfer rate and develop the heat transfer coefficient correlations. The effect of various operating variables like diameter of tube, gap between the concentric coils and coil diameter. The gap between concentric coils and tube diameter affects the heat transfer coefficients and results obtained by them suggests that with increase in coil gap results in the decrease of heat transfer coefficient and when tube diameter increase the heat transfer coefficient increases.

Jayakumar et al.(2010) has done both experimental and numerical analysis so as to find out the local Nusselt number variation along circumference and also along length of helical coil. Variations were made on pitch circle diameter, pipe diameter and tube pitch and how they affect heat transfer rate was found out. In the literature Nusselt number prediction was also done. The variation of Nusselt number corresponding to angular location of point position was also predicted. The conclusion made by them suggest that heat transfer coefficient and therefore along the periphery of helical coil Nusselt number is not uniform and they have derived an expression for calculating the Nusselt number at different points along the periphery of the helical coil in the fully developed region. The effect of pipe diameter was also studied and results suggests that for low pipe diameter, the secondary flow is weak and fluid mixing is less

When the diameter of the coil increases the heat transfer at the outer surface is highest. The PCD influence the centrifugal force of fluid flowing inside the tube, which in turn affects the secondary flow. When the PCD is increased, the curvature effect on flow pattern decreases and the centrifugal force plays a lesser role in flow characteristics.

Heat transfer and flow characteristics in a spiral-coil tube was studied by Naphon (2011).Both experimental and numerical study on horizontal spiral-coil tube was carried out to predict the flow characteristics. To model the turbulence the standard k–ε two equation model was used. He found out that heat transfer coefficient was affected by centrifugal force and also the Nusselt number and pressure drop obtained from the spiral-coil tube are almost one and half times higher than that of straight tube heat exchanger due to the effect of centrifugal force

Pawar et al. (2014) has done the experimental analysis by considering isothermal steady state as well as non-isothermal unsteady state conditions for laminar as well as turbulent flow conditions in helical coils by considering Newtonian as well as non-Newtonian fluid as the working fluid. The Newtonian fluid considered is Glycerol-water mixture (10 and 20 % glycerol) and Non-Newtonian fluid. Considered is 0.5–1% (w/w) dilute aqueous polymer solutions of Sodium Carboxy Methyl Cellulose and Sodium Alginate is considered as Correlation was found out between Nusselt number, coil curvature ratio and Prandtl number.

Lu et al. (2014) has done both numerical and experimental work on shell-side thermal hydraulic performance of multilayer spiral wound heat exchangers subjected to different thermal boundary conditions for the wall.

2.3. NUMERICAL WORKS

Rennie et al. (2006) has done numerical analysis of double-pipe helical heat exchanger .The heat exchanger was numerically investigated for laminar flow conditions for both parallel flow and counter flow configuration and heat transfer rate for different flow rates and tube size was also looked upon. The correlation between annulus Nusselt numbers and a modified Dean number was found.

Huminic et al.(2011) has done numerical investigation on double tube helical coil heat exchangers heat transfer characteristics. Working fluid consider for heat exchanger was Nano fluids. CuO and TiO2 where used as Nano particles in the working fluid. They consider laminar flow condition in the heat exchanger. They suggested that concentration of Nano particle and Dean Number a function of curvature ratio affects the heat transfer rate and heat transfer coefficient in helical coil heat exchanger respectively and with the increase in Dean number the rate of heat transfer increases.

Ferng et al. (2012) has done numerical investigation focussed on predicting the effect of dean number and pitch size of the tube on the hydraulic as well as thermal characteristics of helical coil heat exchanger. Three Dean Numbers and four pitch sizes where considered for the study. The secondary flow in the tube, developing flow, turbulent wake around the rear of coiled tube and heat transfer from entrance region was studied by them.

Jahanmir et al. (2012) has done numerical investigation on shell and tube heat exchanger having single twisted tube bundle for five different twist angles and then compared the result obtained with the conventional shell and tube heat exchanger with single segmental baffles. The effect of shell side nozzles configuration on heat exchanger performance was also studied.

The analysis of results explains that for same shell side flow rate, the heat transfer coefficient of heat exchanger with twisted tube bundle is lower than that of heat exchanger with segmental baffle. Heat exchanger shell side pressure drop with twisted tube bundle is also much lower compared to that of heat exchanger with segmental baffles. For heat exchanger with twisted tube bundle pressure drop reduces rapidly compared to that of single segmental baffle heat exchanger. In the range of 25^o – 65^o twist angle the overall heat transfer coefficient and pressure drop was negligible. The maximum heat transfer rate for a given Pressure drop was corresponding to angles 55^o and 65^o

2.4. EXPERIMENTAL WORKS

Enhancing the heat transfer by introducing helical tapes has been studied by Eiamsa-ard et al. (2005) for straight tube. Experimentally they found that helical tape inserts in the inner tube of heat exchanger enhances the heat transfer rate by introducing swirl motion and the swirl motion induced will increase the turbulence and there by heat transfer.

The effect of screw tapes with or without core rod on friction factor and heat transfer performance has been studied by Eiamsa-ard et al. (2006) experimentally.

Jamshidi et al. (2013) has experimental work on shell and helical tube heat exchanger to enhance heat transfer in helical tube section hot water is flowing and cold water is flowing on the shell side. The determination of heat transfer coefficient was done using Wilson plots. Taguchi method was used to optimise the coil diameter, pitch of the coil and shell side flow rate. He found out that coil diameter of helical coil, coil pitch and tube side flow rate are the most relevant parameters in helical coil heat exchangers. From the experimental works it was found that coil pitch affect the Nusselt number and this caused due to fluid flow rate. The high value for tube side Nusselt number is obtained for lowest coil pitch and high tube side flow rate this is caused due to higher torsion occurring for lower pitches. With the decrease in coil pitch the curvature of tube increases and strong secondary flow is produced in tube side which enhances the heat transfer. The tube side Nusselt number and overall heat transfer coefficient increases with increase in coil diameter of the tube.

After extensive literature review it has been found that although many work has been done on helical coil heat exchanger without baffle to find out the heat transfer coefficient and effect of curvature ratio and pitch of the coil no work has been done comparing Nusselt number for Helical coil with baffle and without baffle.

2.5 GAPS IN LITERATURE

So many work has been done on finding heat transfer coefficient in helical coil heat but no work has been found comparing the variation of Nusselt number for helical coil heat exchanger with and without baffle

2.6 AIMS AND OBJECTIVES

The Main aim of this project is compare the value of Nusselt number with Reynolds number for helical coil heat exchanger with and without baffles

The case under consider are

1.Laminar flow in Helical coil for different curvature ratio and with and without baffles for different flow rates of hot fluid

2.Turbulent flow in Helical coil for different curvature ratio and with and without baffles for different flow rates of hot fluid

CHAPTER 3
PROBLEM FORMULATION

3.1. PROBLEM SPECIFICATION

In my study I have considered helical coil tube in tube heat exchanger with ring baffle. The number of turns considered is two. In the inner helical tube hot fluid is flowing and in the outer helical annular region the cold fluid is flowing the ring baffle is placed in the annular region. Number of baffles placed is three and is placed at equal distance from inlet to the outlet. The coil diameter (D) is varied from 80mm to 200 mm in steps of 40 mm. The thickness of inner pipe and outer pipe is not considered. The pitch of the helical coil is 30 mm. The diameter of the inner helical pipe (d_1) is 8 mm and that of the outer helical pipe diameter (d_2) is 17 mm. The material considered for the pipe of the heat exchanger is copper. The hot and cold fluid considered is water and properties of both copper and water are considered to be constant over the operating range.

In the problem tube diameters are fixed and the coil diameter is varied for both laminar and turbulent flow to study the effect of curvature ratio (d/D).

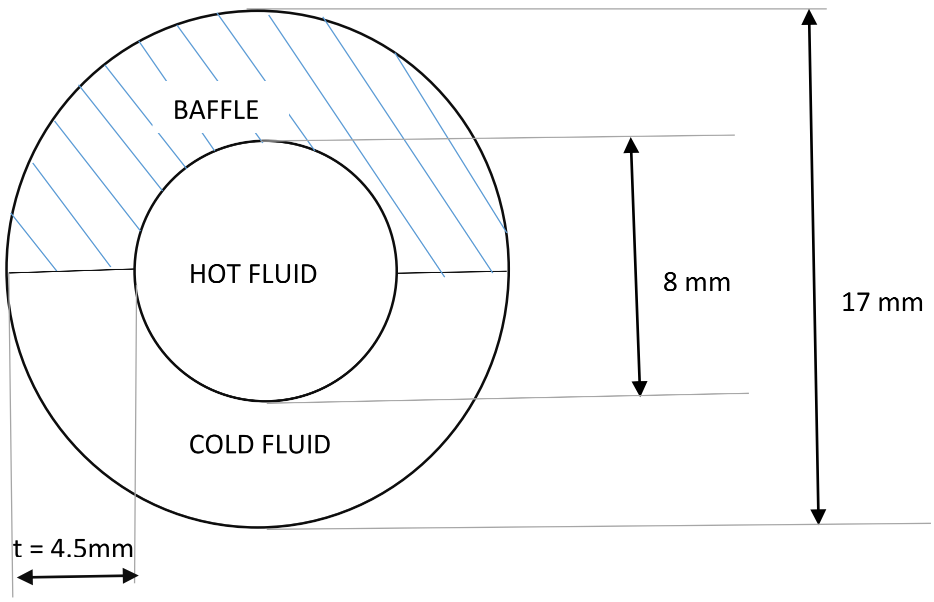


Figure 3.1: Cross section of helical coil tube in tube heat exchanger at the baffle

The thickness of the baffle along the radius in the annular region is varied to study the effect of thickness. The values chosen for the study are t= 3.5, 4, 4.5 mm. and helical coil heat exchanger without baffle is also considered for the study.

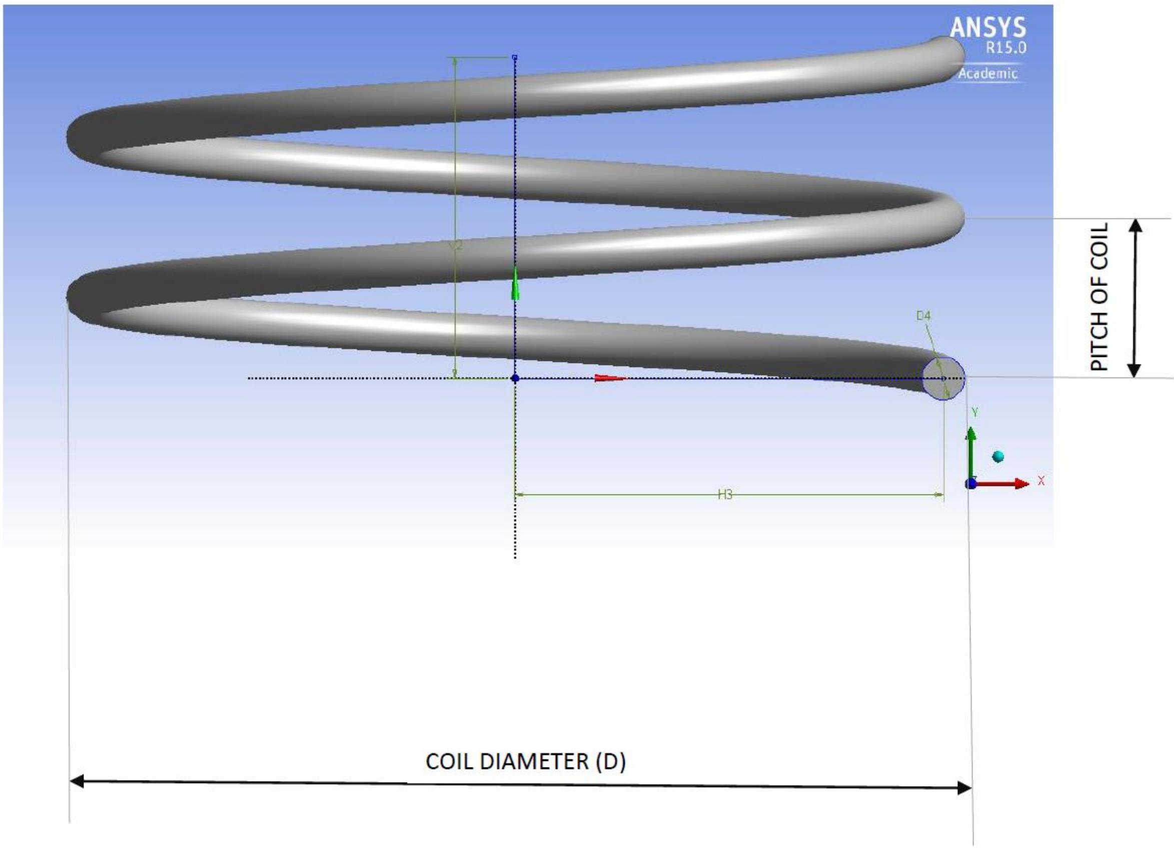


Figure 3.2: Specification of geometry

Table 3.1: Dimensions of helical coil tube in tube heat exchanger

DIMENSIONS	
Coil diameter	variable
Inner tube diameter(d_1)	8 mm
Outer tube diameter(d_2)	17 mm
No: of turns(n)	2
Pitch(H)	30 mm
No: of baffles	3

3.2 CASES CONSIDERED FOR STUDY

Table 3.2: Cases considered for study

SI NO:	CASE NAME	VARIABLES	VALUE RANGE
1	D/d=10 Laminar	Reynolds number and baffle thickness	Re=4000,5000,6000,7000 t (mm)=0,3.5,4,4.5
2	D/d=15 Laminar	Reynolds number and baffle thickness	Re=4000,5000,6000,7000 t (mm)=0,3.5,4,4.5
3	D/d=20Laminar	Reynolds number and baffle thickness	Re=4000,5000,6000,7000 t(mm)=0,3.5,4,4.5
4	D/d=10 Turbulent	Reynolds number and baffle thickness	Re=10000,12000,15000,18000, 21000,24000,25000 t(mm)=0,3.5,4,4.5
5	D/d=15 Turbulent	Reynolds number and baffle thickness	Re=10000,12000,15000,18000, 21000,24000,25000 t(mm)=0,3.5,4,4.5
6	D/d=20 Turbulent	Reynolds number and baffle thickness	Re=10000,12000,15000,18000, 21000,24000,25000 t(mm)=0,3.5,4,4.5

CHAPTER 4
METHODOLOGY

4.1 CREATION OF GEOMETRY

The geometry was created in Ansys workbench Design modeler

4.1.1 Creation of Hot fluid region

Hot fluid is flowing in the inner helical coil. For creating the hot fluid zone the sweep option in the design modeler is used. For using sweep option a profile and a path is required. The profile and path is created in XY plane. The profile here is a circle whose diameter is the diameter of the helical pipe and which is at a distance equal to the radius of the coil from the origin. The path here is a straight line along the Y-axis and whose length is equal to the product of number of turns and pitch. The profile and path are created in different sketches in the XY plane. The helical coil is generated by giving profile, path and number of turns. In the details the type is changed to fluid from solid.

4.1.2 Creation of Baffles

For creating baffles also we are using sweep option. Semi-circular ring baffles are used in the heat exchanger for creating that the cross section of the baffle in created in a new sketch in the XY plane. The newly created sketch act as the profile .Another straight line in a new sketch is created along the Y-axis this will act as the path .Two such line are created, the length of the line is equal to product of number of turn and pitch after doing two sweep operations we will get two solid bodies then Boolean operation is used .The subtract option in Boolean operation is used and the tool body is not preserved. The same procedure is carried out for making the other two baffles also

4.1.3 Creation of Cold fluid region

For creating cold fluid region the outer helical coil is made. Sweep option is used for this, the profile used is circular shape with diameter equal to the diameter of the outer helical pipe and the centre is at a distance which is equal to radius of helical coil from the origin .The cold fluid region is obtained by using the Boolean subtraction, the baffles and the hot fluids region are subtracted from the larger helical coil to obtain the cold fluid region

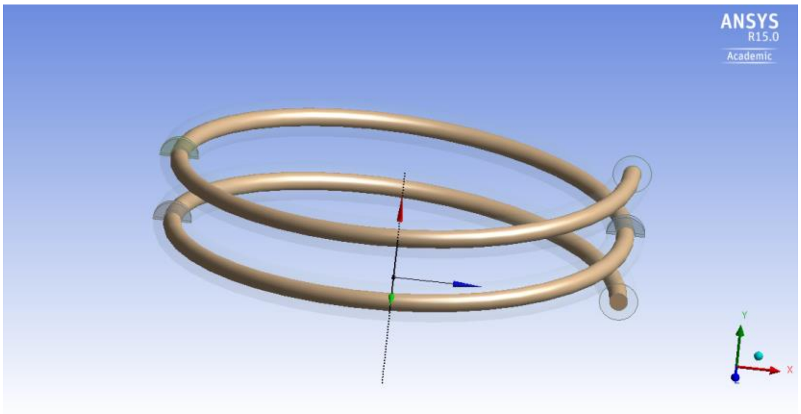


Figure 4.1: Geometry showing hot fluid region, cold fluid region, Baffles

4.2 MESH GENERATION

The geometry should be made sweep able before generating mesh .We can check whether a body is sweep able or not by right clicking on the mesh then go to show option and in that go to sweep able bodies then it will show the sweep able bodies in green colour. Our geometry is not sweep able because it has the imprints of baffles on the hot fluid region and cold fluid region .So now we have to use the slice option (available under create) in design modeler and we should slice the geometry along the planes where the imprints are formed which help us to separate the hot and cold fluid into different small pieces and the body can be made sweep able.

The size of the mesh can be adjusted under the sizing option which is available in details of mesh. We can adjust the min size, max face size and max size to adjust the size of the mesh and the number of elements of the mesh. In the details of the mesh statistics we can check the number of elements and number of node, Element quality, orthogonal quality, skewness whose values are to be within a specified range in order to obtain a good mesh.

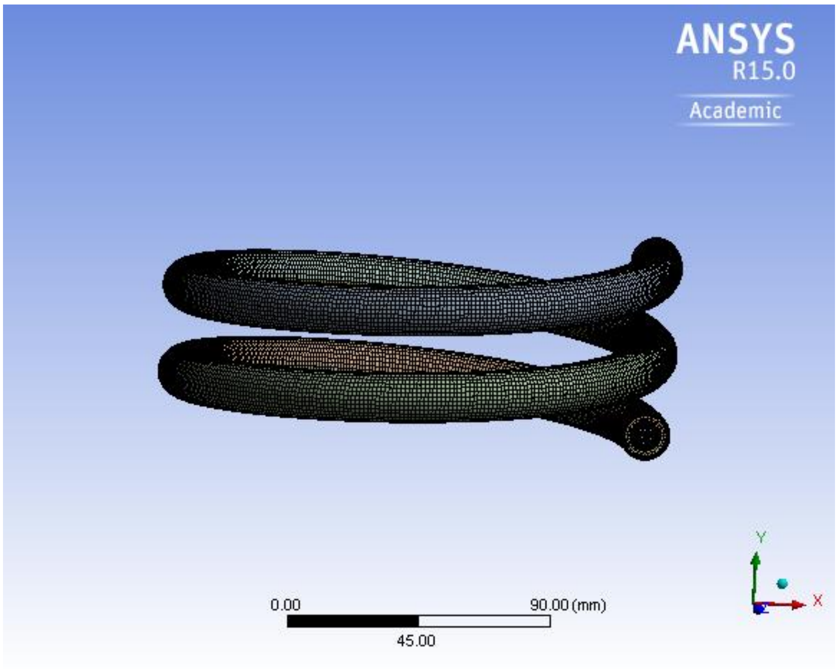


Figure 4.2: Image of geometry which is meshed fully

To apply inflation layer in hot and cold fluid region we have to use inflation option select geometry as hot fluid and boundary as hot fluid wall. In the same way in the cold fluid region inflation layer can be provided

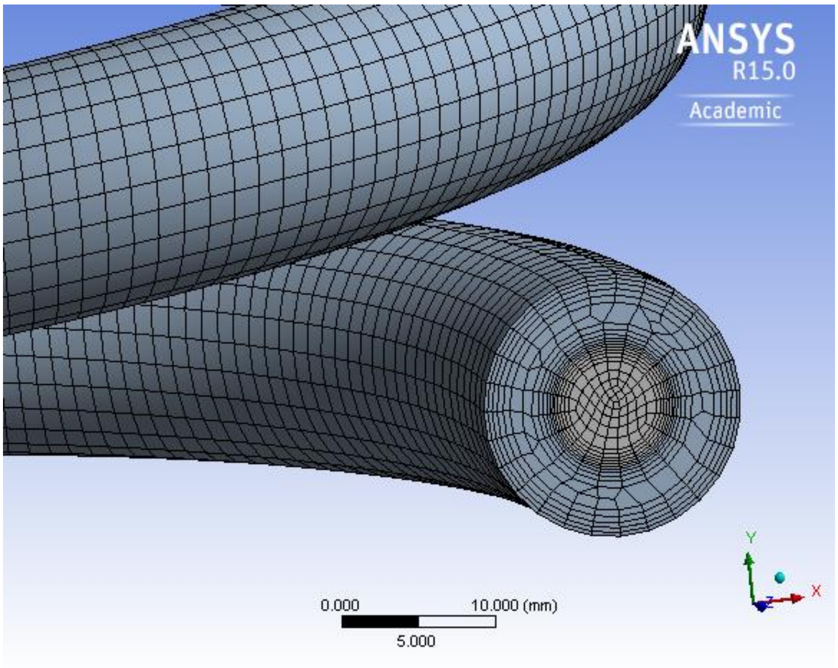


Figure 4.3: Image showing the inflation layer

4.3 BASIC ASSUMPTIONS

1. Steady state heat transfer conditions where assumed.
2. Counter configuration is considered for heat exchangers.
3. Conjugate heat transfer between two fluids considered.
4. Natural convection and radiation was neglected.
5. Laminar and turbulent flow cases are considered

4.4 GOVERNING DIFFERENTIAL EQUATIONS

The Continuity Equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = 0$$

The Momentum Equation

$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v}\vec{v}) = -\nabla p + \nabla \cdot \left[\mu \left(\nabla \vec{v} + \nabla \vec{v}^{T} \right) \right] + \rho \vec{g} + \vec{F}$$

Energy equation

$$\rho c_p \left(\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z} \right) = k\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + \Phi$$

Where Φ is viscous dissipation factor

$$\Phi = 2\mu\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2 + \frac{1}{2}\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)^2 + \frac{1}{2}\left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)^2 + \frac{1}{2}\left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z}\right)^2 \right]$$

Turbulence presence in the domain has been modeled using standard k-ε model

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho\varepsilon - Y_M + S_k$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} + S_\varepsilon$$

G_k= Turbulence kinetic energy generation as a result of the mean velocity gradients.

G_b= Turbulence kinetic energy generation as a result of buoyancy.

Y_M= fluctuating dilatation contribution in compressible turbulence to the overall dissipation rate.

$C_{2\varepsilon}$, $C_{3\varepsilon}$ = Constants.

S_k & S_e = The User-defined source terms.

$\mu_t = \rho C_\mu . \frac{k^2}{\varepsilon}$ = Turbulent viscosity

Critical Reynolds number as per the Schimidt correlation (1967)

$Re_{cr} = 2300[1 + 8.6\ (d/D)^{0.45}\ ]$

The heat transfer coefficient can be obtained by the following relation

$$h = \frac{-k\frac{\partial T}{\partial x}}{T_W - T_f}$$

Local Nusselt number is given by

$$Nu_x = \frac{hl}{k}$$

Another representation is as follows

Local Nusselt number is given by

$$Nu_x = \frac{-\frac{\partial T}{\partial x}\,\mathrm{dh}}{T_W - T_f}$$

Then the average Nusselt number can be found by following relation

$$Nu_{avg} = \frac{1}{L}\int_0^L Nu_x dx$$

Reynolds number value is given by formula

$$Re = \frac{\rho V D}{\mu}$$

4.5. VALUES OF PARAMETERS USED

Table 4.1: Properties of Water

DESCRIPTION	VALUE	UNITS
VISCOSITY	0.001003	kg/m-s
DENSITY	998.2	kg/m^3
SPECIFIC HEAT CAPACITY	4182	J/kg-K
THERMAL CONDUCTIVITY	0.6	W/m-K

Table 4.2 Properties of Copper

DESCRIPTION	VALUE	UNITS
DENSITY	8978	kg/m^3
SPECIFIC HEAT CAPACITY	381	J/kg-K
THERMAL CONDUCTIVITY	387.6	W/m-K

4.6 BOUNDARY CONDITIONS USED

The counter flow configuration is considered for helical coil heat exchanger.

The following are the boundary conditions considered.

The flow rate of cold fluid has been kept constant. For turbulent flow the Reynolds number of cold fluid was taken as Re=25000 and for laminar flow the Reynolds number of flow was taken as Re=7000

The flow rate of hot fluid was varied. The flow rate of hot fluid for turbulent flow are varied as flows Re = 10000,12000,15000,18000,21000,24000,25000 and flow rate of hot fluid for laminar flow are Re = 4000,5000,6000,7000. The velocity of hot fluid and cold fluid is specified using method Component and for turbulent flow it is as follows For hot fluid U_z = 1.25601, 1.55072, 1.884, 2.26081, 2.6376, 3.01442, 3.14002 m/s

For cold fluid U_z = -3.14002 m/s

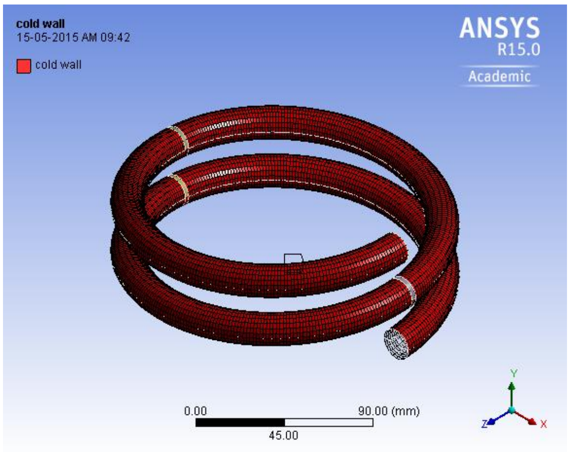
For laminar flow the velocities are as flows

For hot fluid U_z = 0.502404, 0.6280054, 0.75360649, 0.8792075 m/s

For cold fluid U_z = -0.781517843 m/s

The inlet temperature of hot fluid is 355 K and inlet temperature of cold fluid is 290 K

The cold wall is considered to be at constant temperature T= 330K



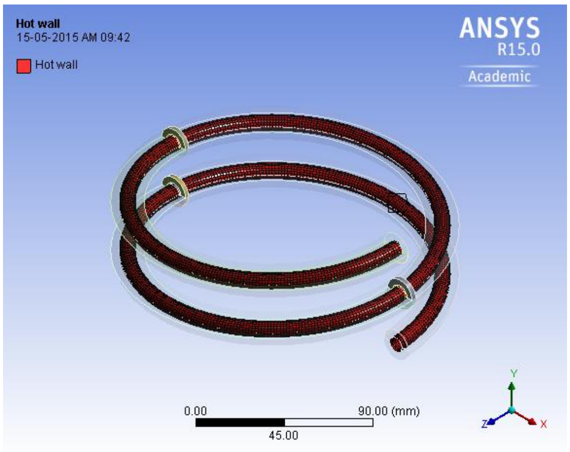


Figure 4.4: Various boundary conditions showing hot and cold fluid walls

CHAPTER 5
RESULTS & DISCUSSIONS

5.1 GRID INDEPENDENCE TEST

The grid independence test is very important and should be done because the final result should be independent of the number of grid. In numerical simulation results depends on the number of grids generated up to a certain limit but according to our need result should not change with the change in number of grid. The grid independence test was done for all the cases to ensure that the final results are independent of number of grids.

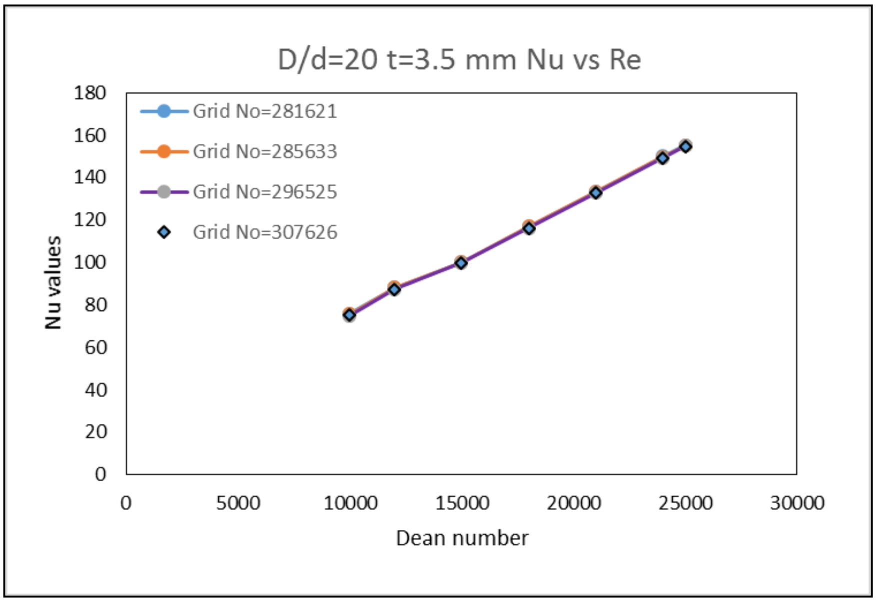


Figure 5.1: Grid Independence test done for D/d=20 with baffle

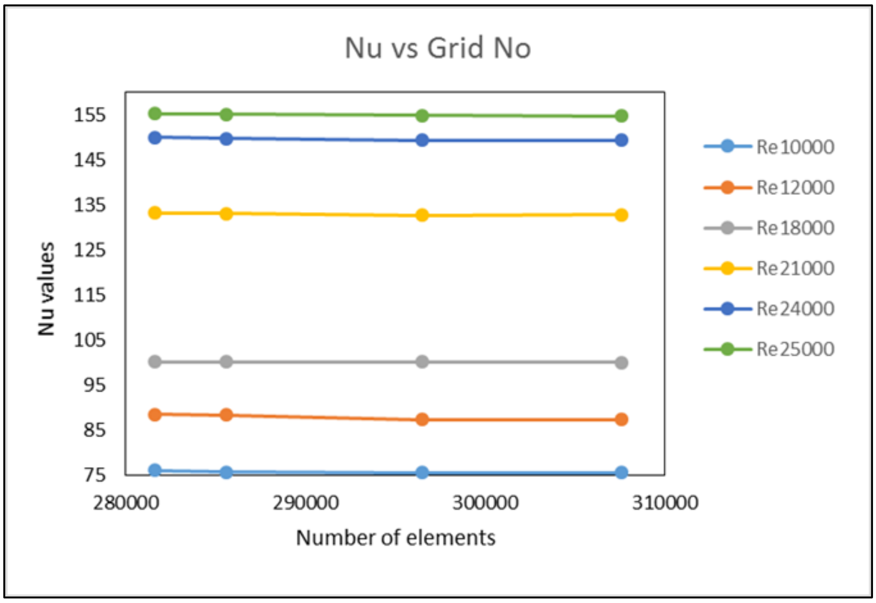


Figure 5.2: Variation of Nu with the Number of elements for D/d=20 in case of laminar flow

The graph shown above clearly indicates that after number of elements=296525 there is not much variation in Nusselt number value so it is clear from the graph that for Number of elements greater than this value the result will be independent of the number of elements.

5.2 Variation of Nusselt number with Reynolds number for different curvature ratio

5.2.1 Laminar case

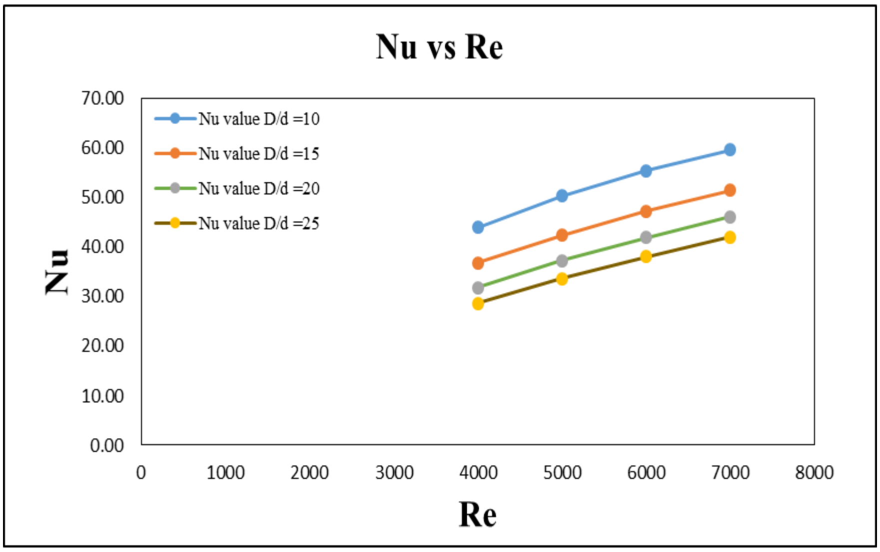


Figure 5.3: Variation of Nu with Re for different D/d ratio for laminar flow

With the increase in value of D/d ratio the value of average Nusselt number is decreasing that is for a particular value of Re the value of Nu will be maximum for D/d = 10 as the value of D/d ratio increases the effect of curvature decreases and the action of the centrifugal force reduces and that is the reason for the decrease in the Nu value. The effect of curvature is the main reason for increase of centrifugal force as the centrifugal force increases it will accelerate the particle and thus results in the increase in velocity of fluid in the helical coil heat exchanger. As the Re increases the value of Nu increases for a particular D/d ratio. The more velocity of the flow will be toward the outer walls in case of helical coil and for less D/d ratio the secondary flow which is the main mechanism of heat transfer in helical coil heat exchanger is also enhanced. As the coil diameter increases the behaviour of Helical coil is tending to that of a straight tube heat exchanger.

5.2.2 Turbulent case

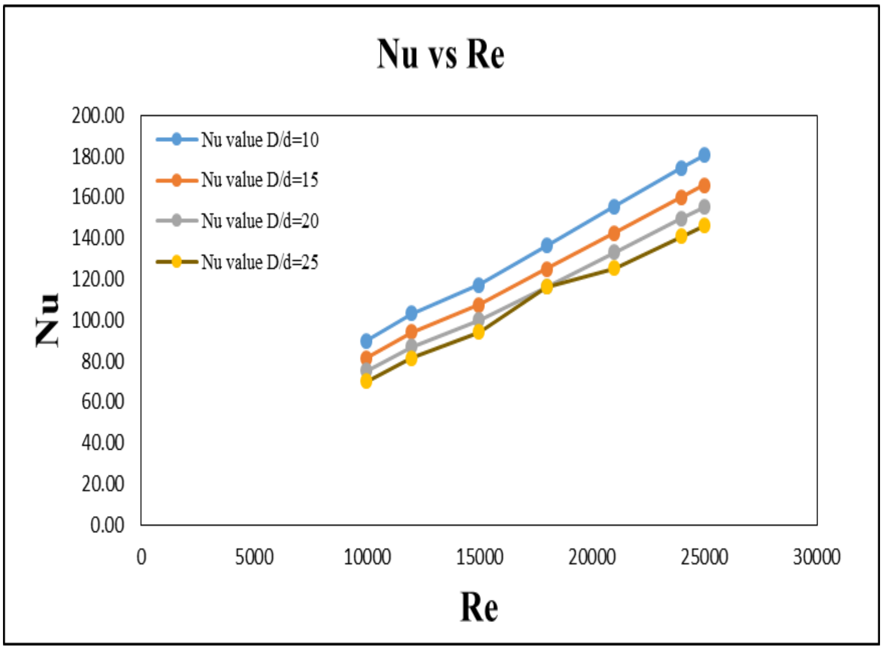
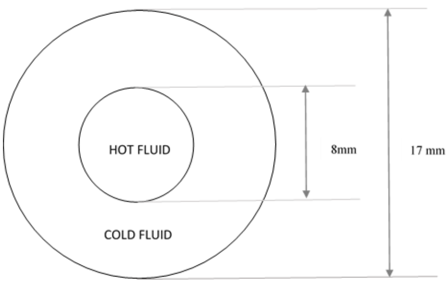
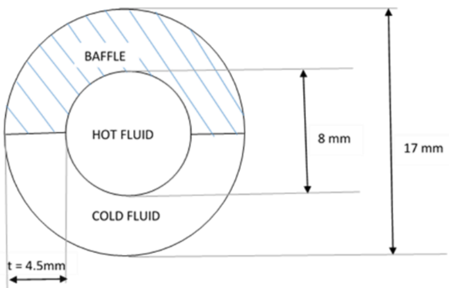


Figure 5.4: Variation of Nu with Re for different D/d ratio for turbulent flow

For turbulent flow also with the increase in D/d ratio the Nu value is decreasing for the same flow rate of the hot fluid as in the case of laminar flow. The reason for reduction in Nusselt number can be explained in the same way as explained for laminar case.

5.3 VARIATION OF NUSSELT NUMBER WITH DIFFERENT THICKNESS OF BAFFLE





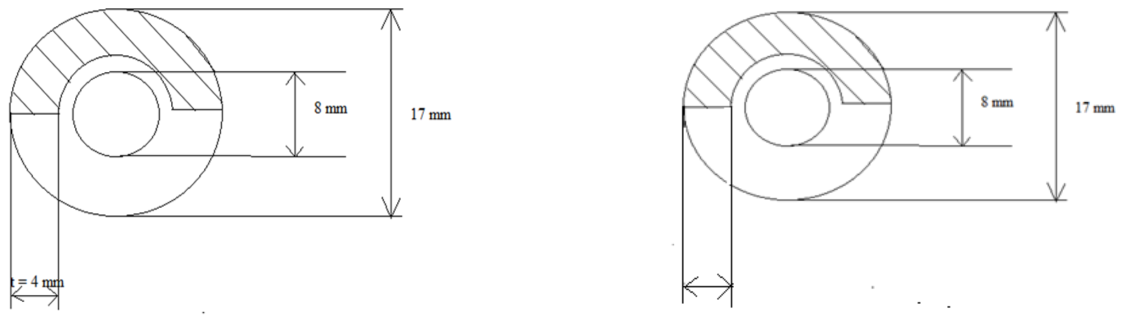


Figure 5.5: Schematic showing the variation in thickness of baffle.

5.3.1. LAMINAR CASES

5.3.1.1. D/d =10

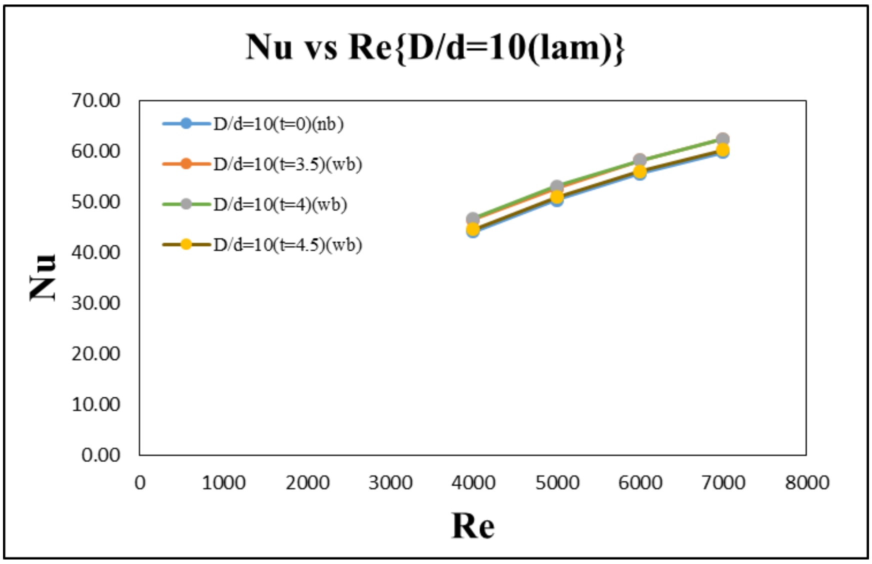


Figure 5.6: Variation of Nu with Re for D/d=10 Laminar flow with different thickness of baffle

The graph above shows the variation of Nu for different Re for hot fluid. The graph shows there is an increase in the Nu value for the case considered with baffle to that without baffle.

The Nu values are increasing with the increase in Re values because with the increase in Re value the velocity of the fluid is increasing and increase in velocity helps in increasing the convection or in other words it will reduce the convective resistance which results in high Nu values.

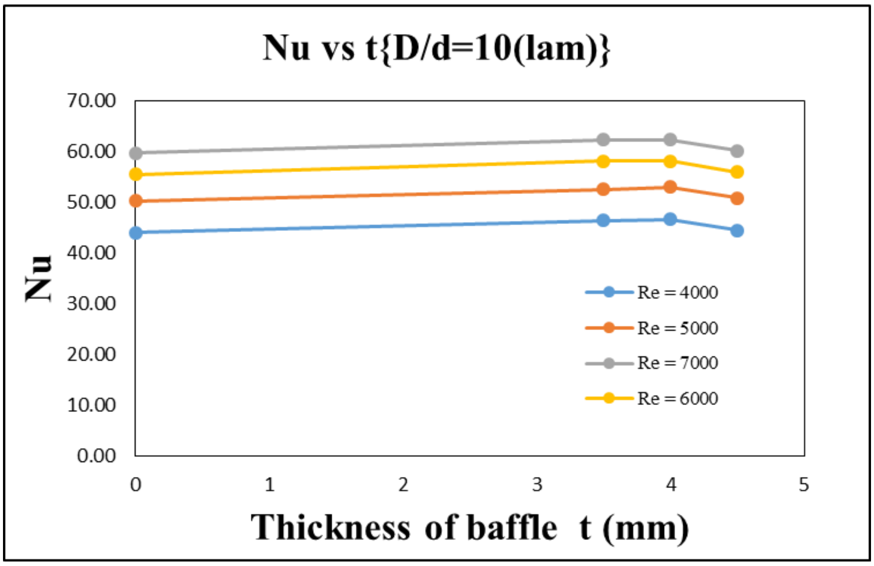


Figure 5.7: Variation of Nu with t for D/d=10 Laminar flow for different Re values

The result shown above clearly indicates that with decrease in thickness of baffle the Nu value is increasing. The lowest value of average Nusselt number is obtained for the case without baffle. For thickness of 3.5mm and 4 mm the value of Nu is more compared to that with no baffle. For baffle thickness of 4.5 mm the value of average Nusselt number is decreasing compared to that with t = 3.5mm & 4mm.

The reduction in Nu value for t = 4.5 mm can be due to the following reason. For t = 4.5 mm the contact area of cold fluid with hot fluid walls is little reduced because the baffle is occupying that area and as the area reduces the convection also reduces which results in low value of Nu. In case of t=3.5 mm and t=4mm there is a gap between the baffle and the hot fluid wall and this gap is very small when flow of cold fluid occurs through this narrow gap the velocity of the flow near the hot fluid wall increases and as a result the convection increases.

In case of t = 4.5 mm there is no such gap between the baffle and the hot fluid wall so there will not be any increase of velocity near the baffle so convective heat transfer is not enhanced near the hot fluid wall and this is one of the reason behind decrease in Nu number

5.3.1.2. D/d = 15

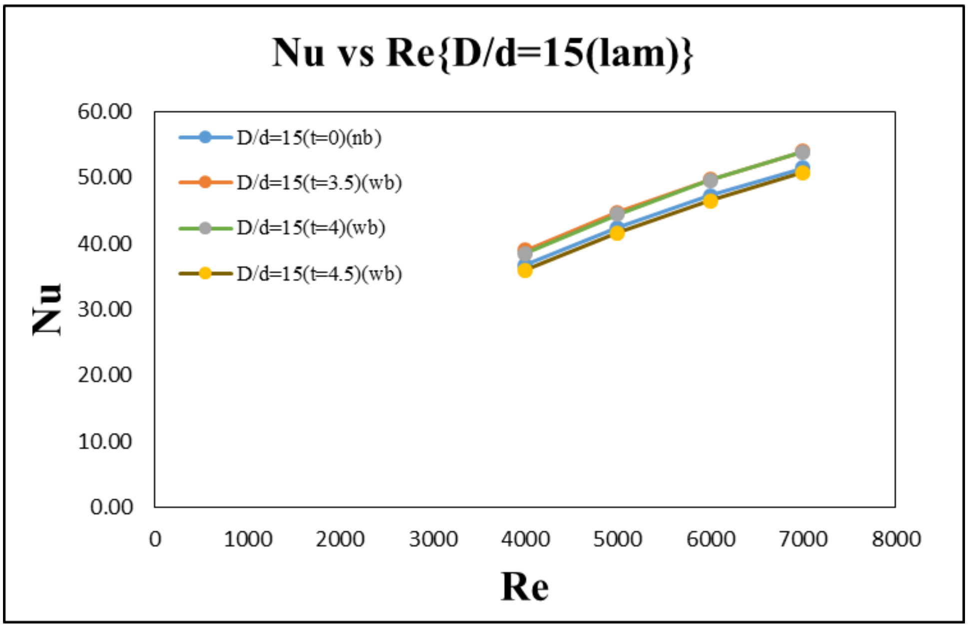


Figure 5.8: Variation of Nu with Re for D/d=15 Laminar flow with different thickness of baffle

The figure above shows the variation of Nu with Re for different thickness of baffle. The Nu value increases with increase in Re value because with the increase in the velocity of the flow convection increases because velocity is an important factor which affects convection.

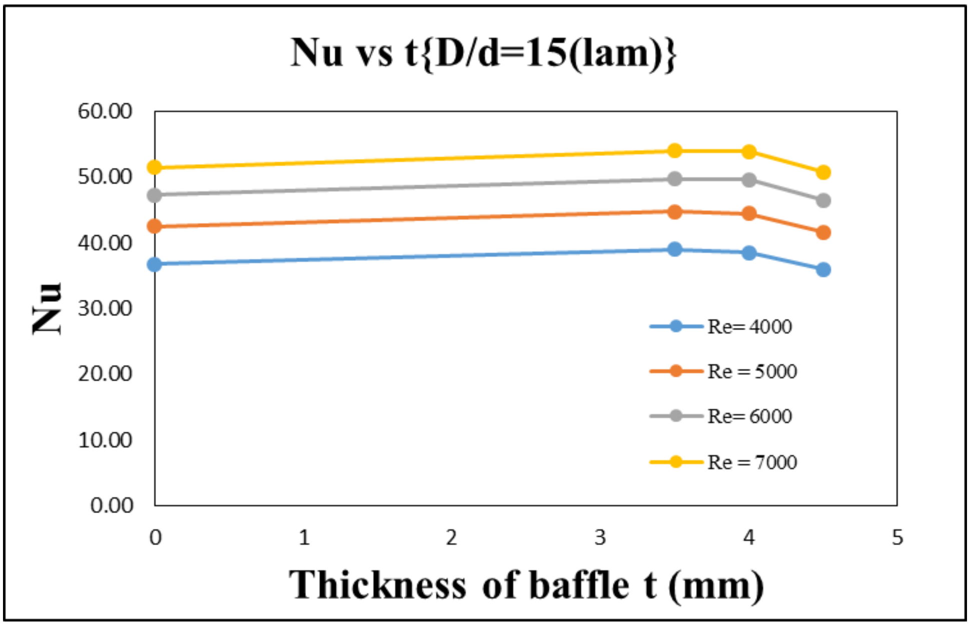


Figure 5.9: Variation of Nu with different thickness of baffle for D/d=15 Laminar flow for different values of Re

This case also follows the same trend as that of D/d = 10 that is with the reduction in baffle thickness the Nu value is increasing. This case shows that for moderate values of D/d ratio the variation of Nu with Re is same as that for low values of D/d and proves that curvature ratio does not have any effect.

5.3.1.3. D/d=20

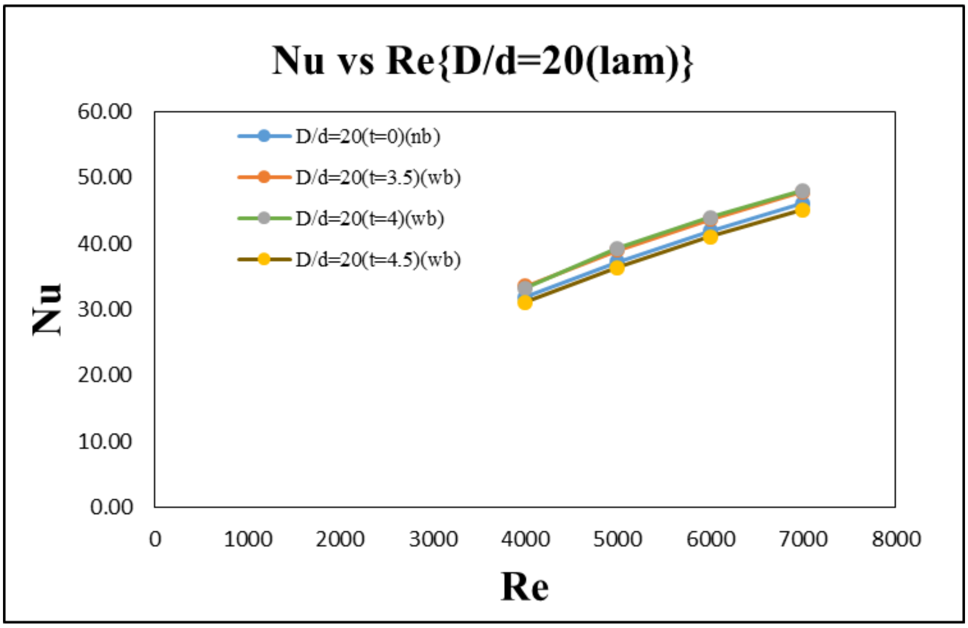


Figure 5.10: Variation of Nu with Re for D/d=15 Laminar flow with different thickness of baffle

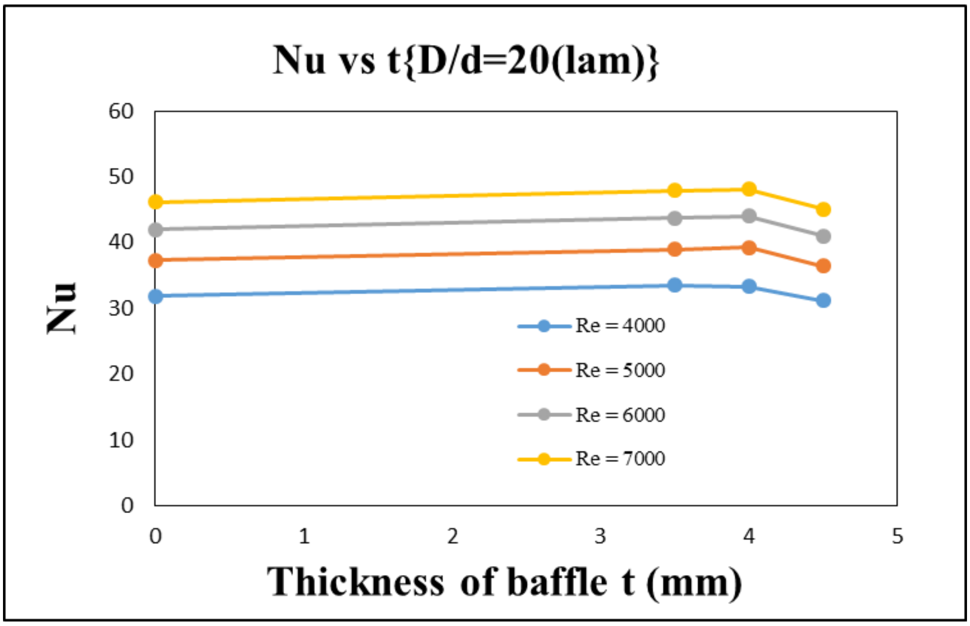


Figure 5.11: Variation of Nu with different thickness of baffle for D/d=15 Laminar flow

This case also follows the same trend as that of D/d = 10 and D/d = 15 that is with the reduction in baffle thickness the Nu value is increasing. This case shows that for high value of D/d ratio the variation of Nu with Re is same as that for low values and moderate values of D/d and proves that curvature ratio does not have any effect.

5.3.2. TURBULENT CASE

5.3.2.1. D/d = 10

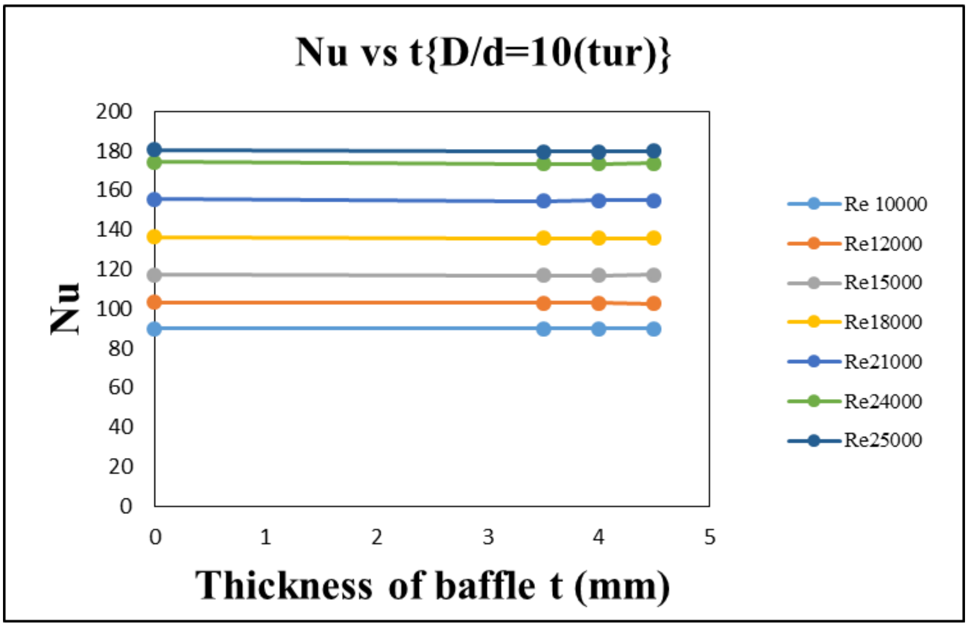


Figure 5.12: Variation of Nu with thickness of baffle t (mm) for D/d=10 turbulent flow for various Re values

For the turbulent case with the introduction of baffle the value of Nu is increasing slightly for moderate values of Re. For low values of Re the Nu value is slightly decreasing for baffle thickness of t = 4.5 mm and for high values of Re the Nu values are slightly increasing.

The percentage decrease of Nu for Re = 10000 is 0.238% and percentage increase in Nu for Re = 25000 is 0.4508%

The results obtained shows that for low Re values the baffle with thickness t = 4.5 mm is obstructing the flow that is why the Nu values are decreasing slightly and for large values the baffle helps to enhance the turbulence and that is why the Nu values are increasing.

The increase and decrease of Nu values with large and small Re value for small coil diameters are very small compared to that for medium size of coil diameter this shows that the effect of introducing baffle is more for medium size of coil diameter in case of turbulent flow were as for laminar flow it is independent of the coil diameter

5.3.2.2. D/d = 15

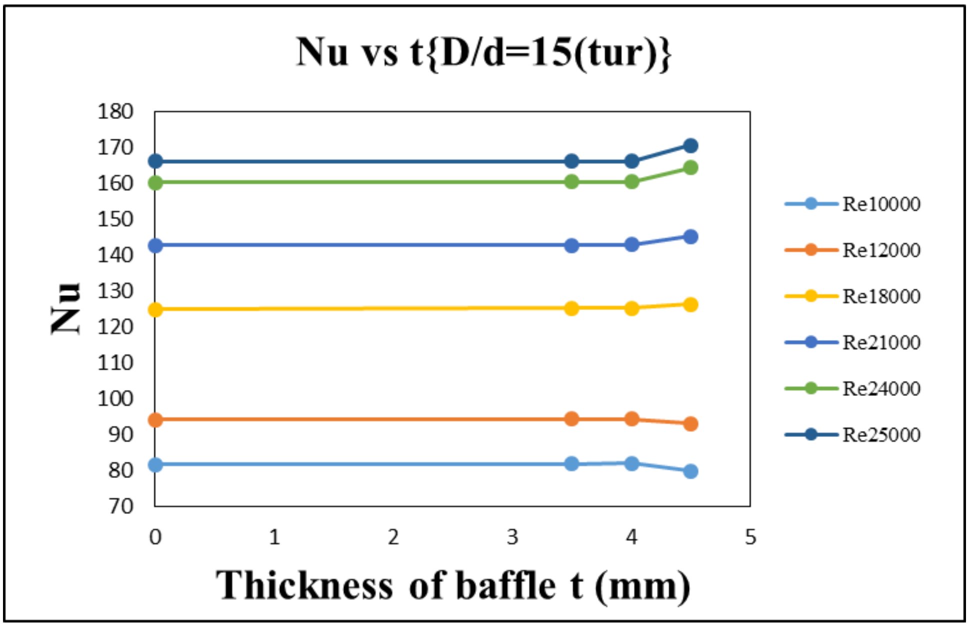


Figure 5.13: Variation of Nu with thickness of baffle t (mm) for D/d=15 turbulent flow for various Re values

The percentage decrease in Nu value for Re = 10000 is 2.254% and for Re = 12000 is 1.135% the value of reduction in Nu is decreasing with increase in Re. For high Reynolds number the value of Nu is increasing with increase in baffle thickness. The percentage increase in Nu value for Re = 12000 is 2.627%

5.3.2.3 D/d = 20

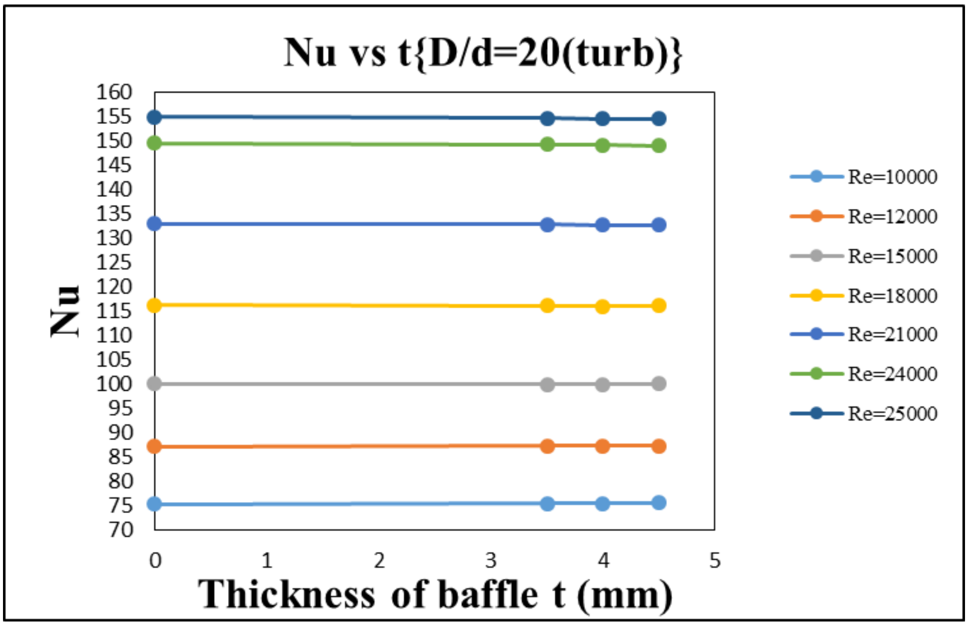


Figure 5.14: Variation of Nu with thickness of baffle t (mm) for D/d= 20 turbulent flow for various Re values

The percentage increase of Nu value for Re =10000 for t= 4 mm and t = 4.5 mm is 0.085 % and between no baffle and t = 4.5mm is 0.227 % and with the introduction of baffle the value of Nusselt number is increasing for Re = 10000.

For Re = 25000 the nu values are decreasing with the introduction of baffle. The percentage decrease for no baffle and t = 4.5 mm is 0.3219 % and percentage increase for t = 4 mm and t =4.5 mm is 0.0536% which is a very less values this shows that with variation in thickness of baffle there is no much variation in Nu value.

5.4. VARIATION OF FRICTION FACTOR WITH Re FOR VARIOUS CASES

The variation of Fanning's friction factor for different Reynolds number is plotted.

5.4.1. LAMINAR CASE

5.4.1.1. For no baffle

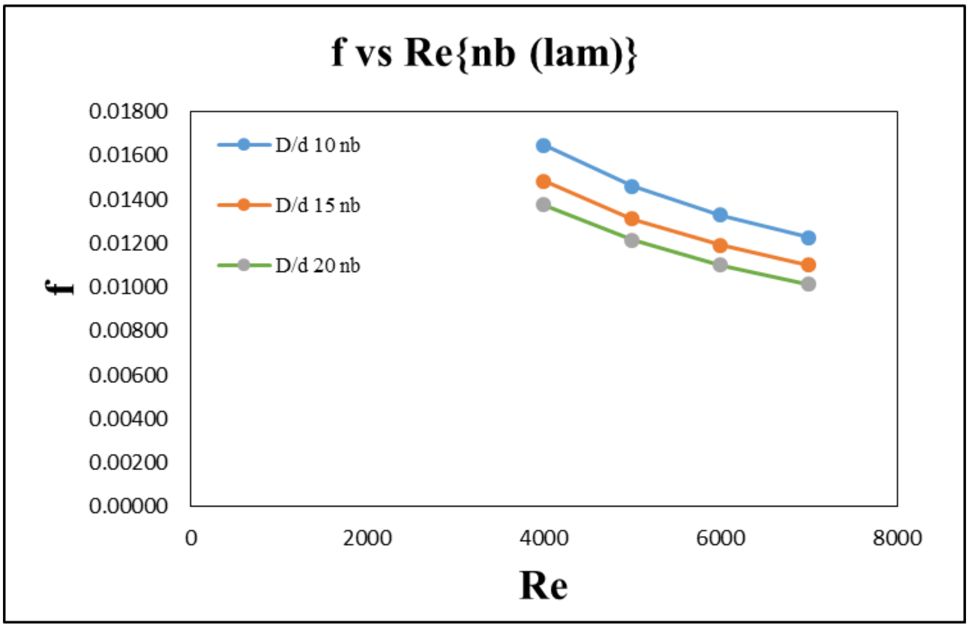


Figure 5.15: Variation of Fanning's friction factor with Reynolds number for laminar flow for different D/d ratios for no baffle

The Fanning's friction factor variation with Re is plotted for different D/d ratios for laminar flow. The maximum values of Fannings friction factor occurs for D/d equal to 10. The Fanning's friction factor inversely varies with the Reynolds number. Fannings friction factor is a strong function of Reynolds number as well as relative roughness of coil surfaces. With D/d ratio increase the surface area increases and relative roughness of coil also increases and roughness dominates the Reynolds number.

5.4.1.2. For baffle thickness t = 3.5 mm

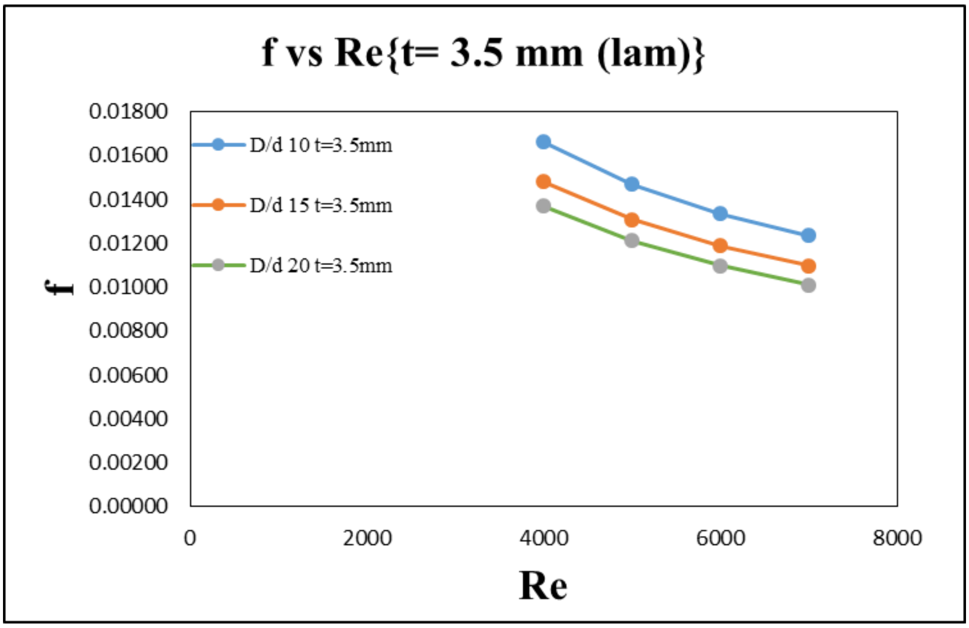


Figure 5.16: Variation of Fanning's friction factor with Reynolds number for laminar flow and for different D/d ratios for t = 3.5 mm

5.4.1.3. For baffle thickness t = 4mm

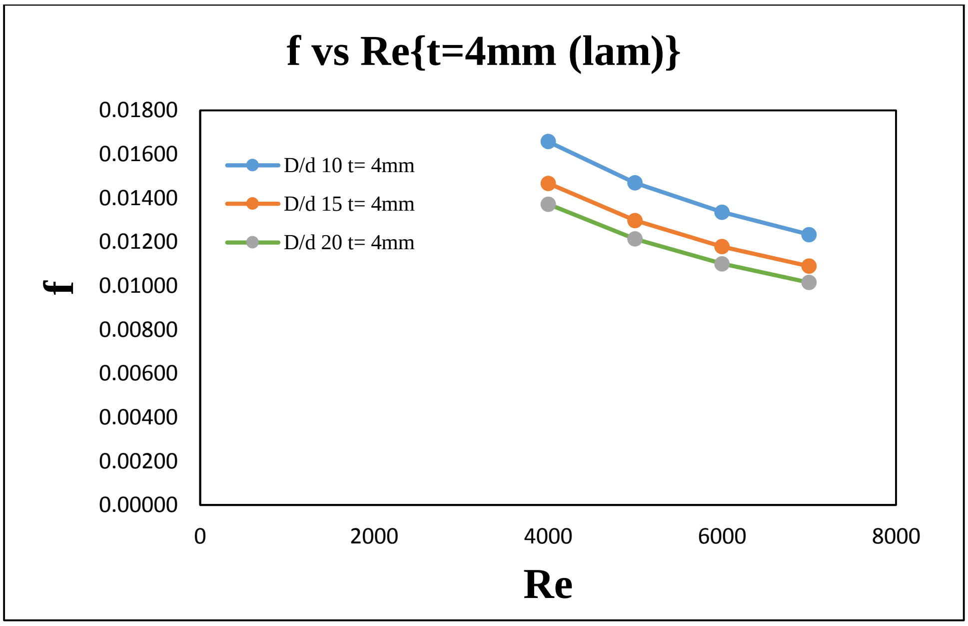


Figure 5.17: Variation of Fanning's friction factor with Reynolds number for laminar flow and for different D/d ratios for t = 4mm

5.4.1.4. For baffle thickness t = 4.5 mm

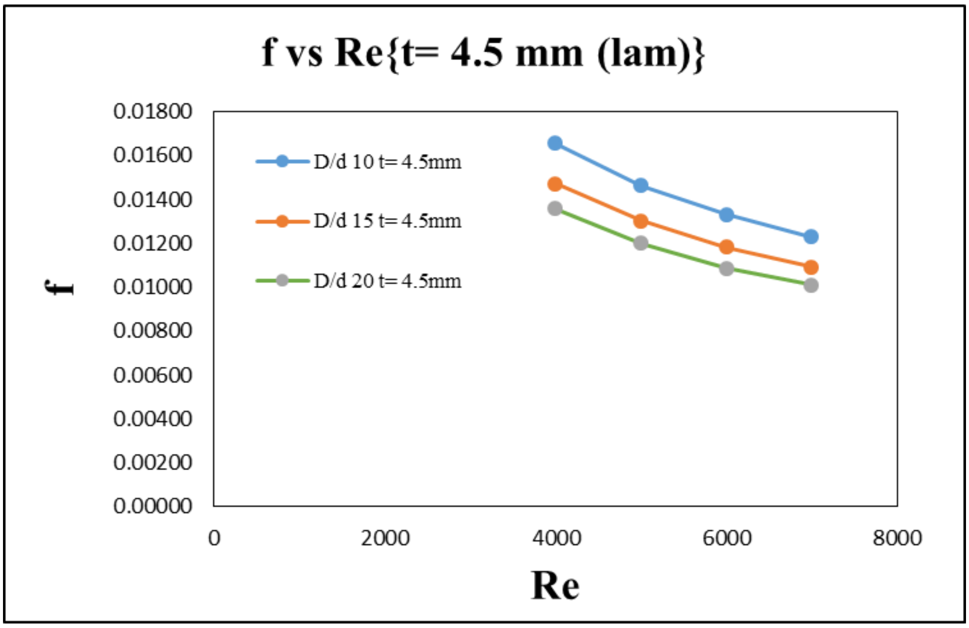


Figure 5.18: Variation of Fanning's friction factor with Reynolds number for laminar flow and for different D/d ratios for t = 4.5 mm

5.4.2. TURBULENT CASE

5.4.2.1. CASE 1: For no baffle

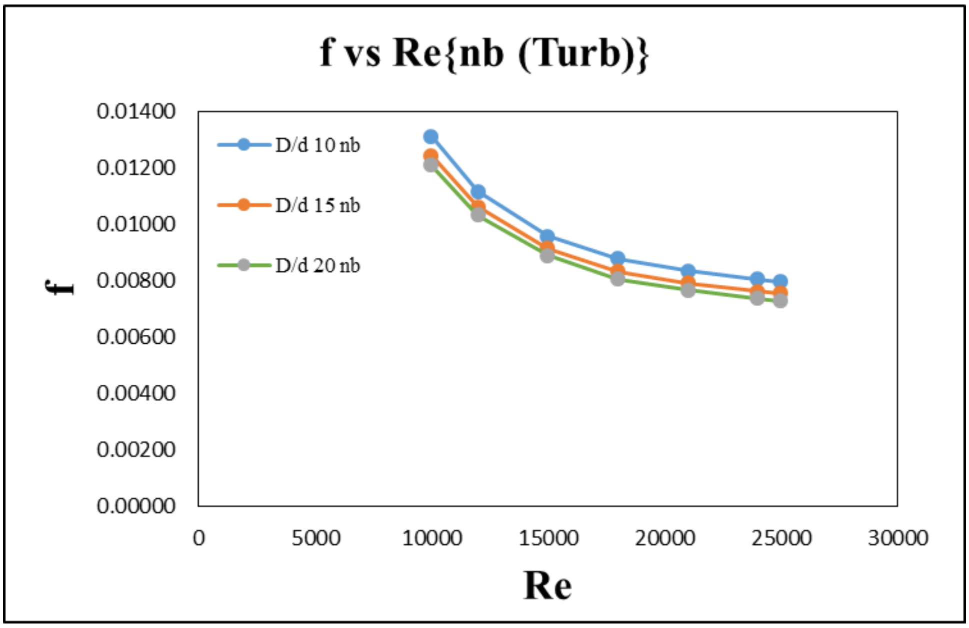


Figure 5.19: Variation of Fanning's friction factor with Reynolds number for turbulent flow and for different D/d ratios for no baffle

5.4.2.2. For baffle thickness of t = 3.5 mm

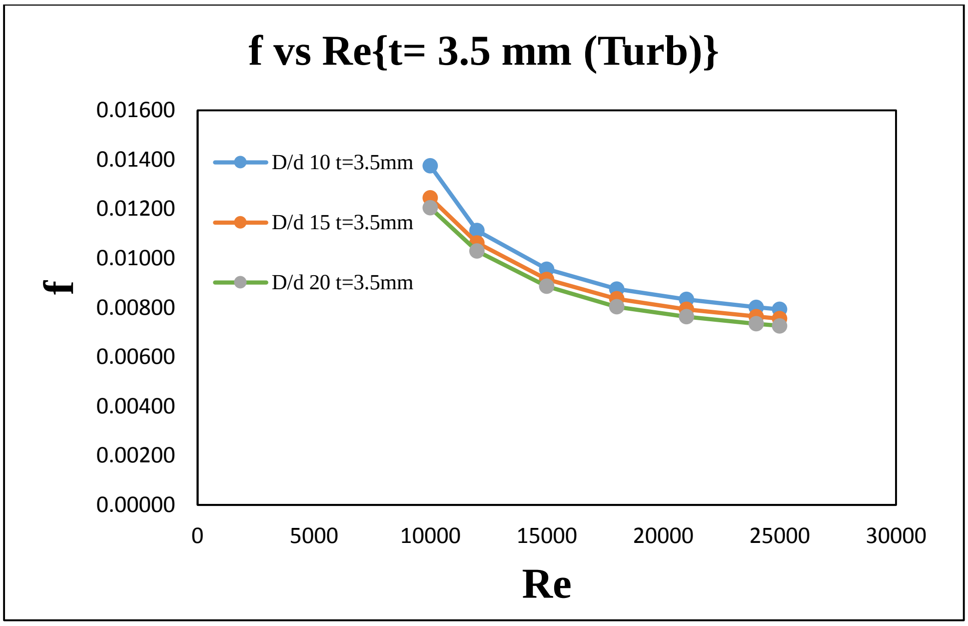


Figure 5.20: Variation of Fanning's friction factor with Reynolds number for turbulent flow and for different D/d ratio for t = 3.5 mm

5.4.2.3. For baffle thickness of t = 4 mm

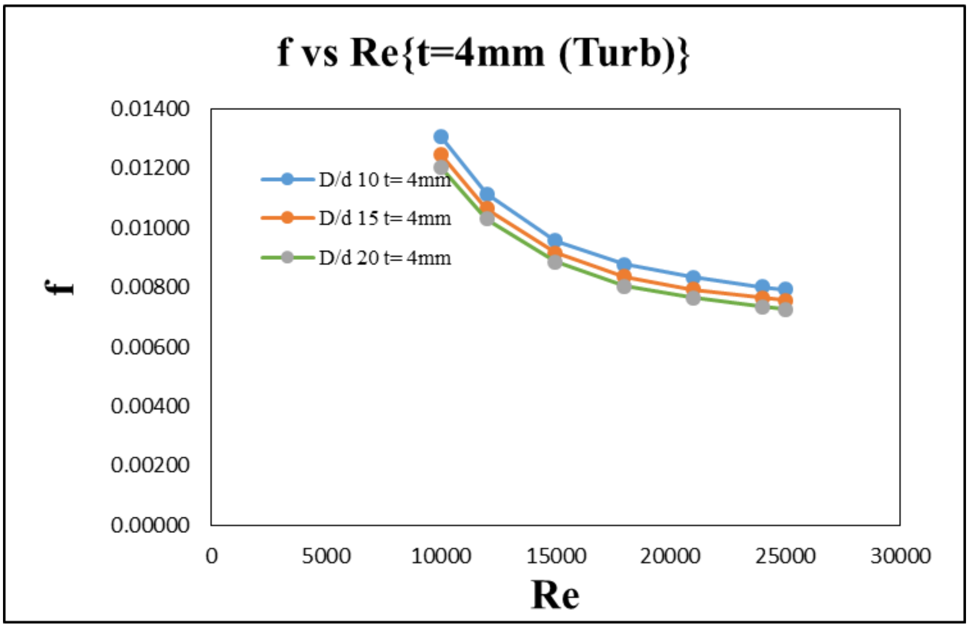


Figure 5.21: Variation of Fanning's friction factor with Reynolds number for turbulent flow and for different D/d ratios for t = 4 mm

5.4.2.4. For baffle thickness of t = 4.5 mm

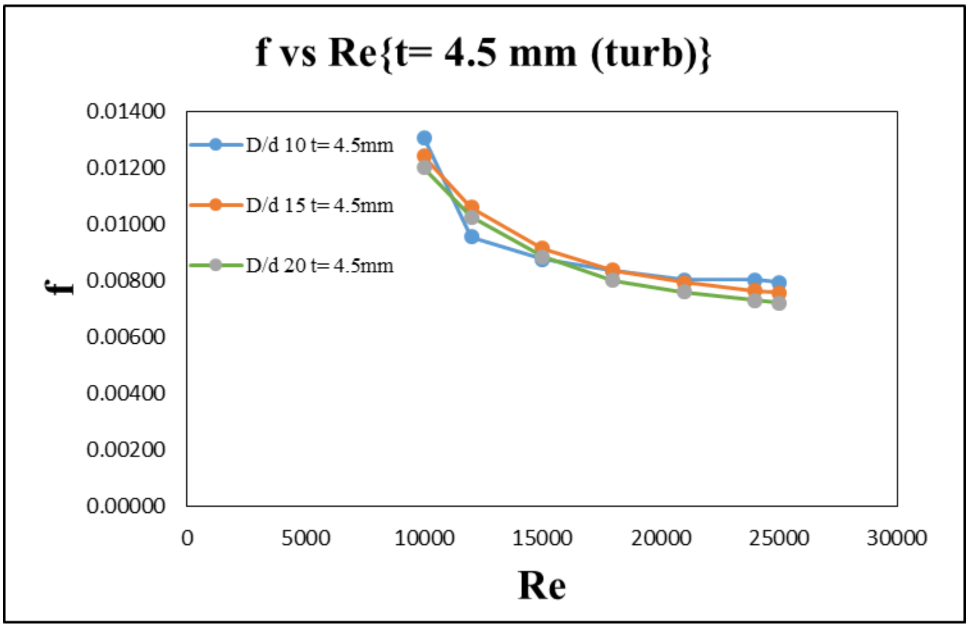


Figure 5.22: Variation of Fanning's friction factor with Reynolds number for turbulent flow and for different D/d ratios for t = 4.5 mm

5.5. VELOCITY CONTOURS OF OUTLET COLD FLUID FOR LAMINAR FLOW

5.5.1. LAMINAR CASE

Re =4000 D/d =10

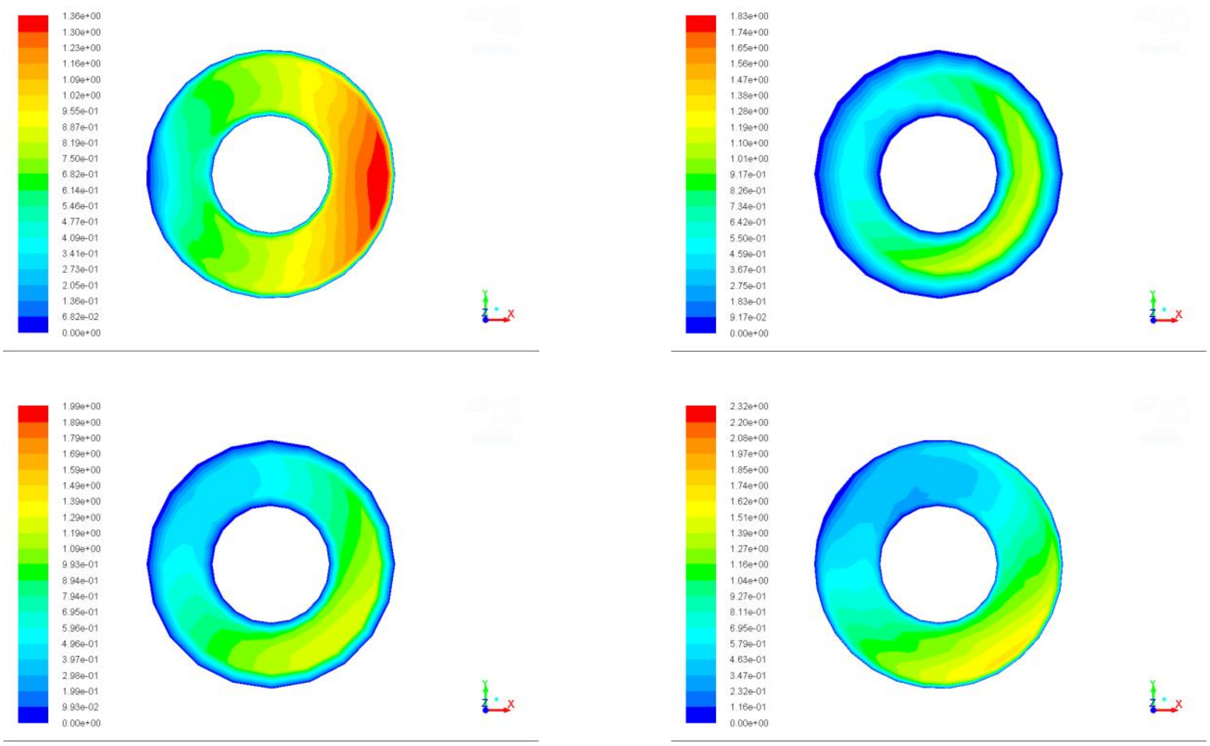
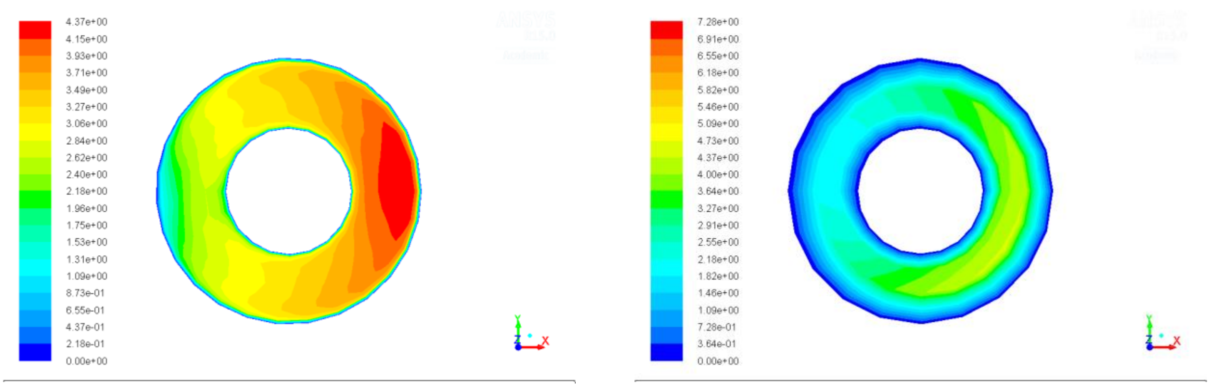


Figure 5.23: Velocity contour for cold fluid outlet in the increasing order of thickness of baffle

5.5.2. TURBULENT CASE

Re =1000 D/d=10



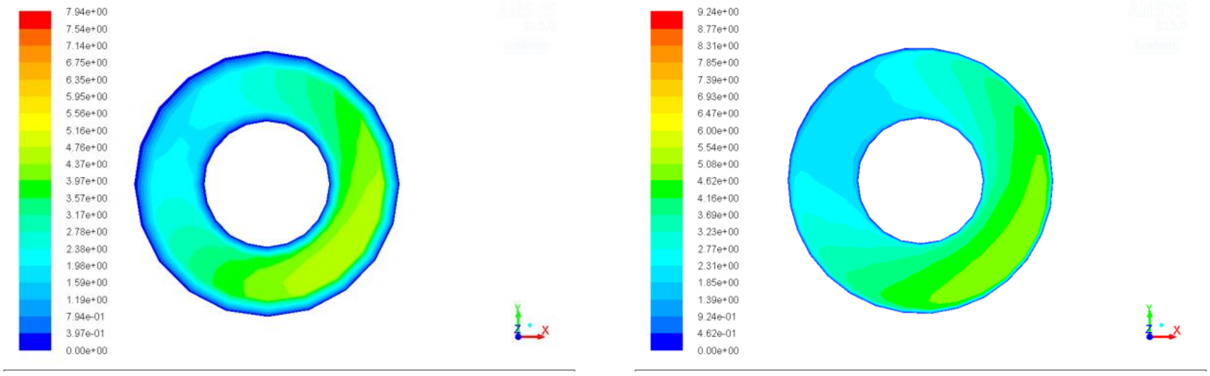


Figure 5.24: Velocity contour for cold fluid outlet in the increasing order of thickness of baffle

CHAPTER 6

CONCLUSIONS & FUTURE SCOPE

6.1 CONCLUSIONS

Numerical simulation of helical coil tube in tube heat exchanger has been done with Ansys fluent and the variation of Nusselt number with different baffle thickness and for various D/d ratio and different flow rate of hot fluid has been plotted.

The conclusion drawn are as follows

- With increase in D/d ratio the Nusselt number is decreasing, the Nusselt number is maximum for D/d=10 for a particular value of Re this is due to the effect of centrifugal force which is more for small D/d ratios and for high D/d ratio the behavior of helical coil tends to that of straight tube.
- For laminar case with baffle thickness of t=3.5 mm and t=4 mm the Nusselt number is increasing compared to case without baffle and thickness of t=4.5 mm.
- For Laminar flow for different D/d ratios the Nu variation with Re follow the same pattern.
- For turbulent case with baffle thickness of t=3.5 mm and 4mm the Nusselt number is slightly increasing and for thickness of t=4.5mm for low Reynolds number the value of Nu is decreasing and for high Reynolds number Nu value is increasing.

6.2 FUTURE SCOPE

Along with numerical simulation experimental validation can also be done to check whether the results obtained are correct or not.

With the study we were not able to finalize the thickness of the baffle .The thickness of baffle can be optimized for obtaining maximum value of Nusselt number

Now the shape of the baffle is ring shape this shape can be modified. The shape of baffle can be optimized so that less obstruct will occur for the flow

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